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THE EFFECT OF CENTRIFUGAL FORCE
ON THE FORCED CONVECTION HEAT TRANSFER COEFFICIENT

A Thesis
Submitted to the Graduate Faculty of the
University of Minnesota

by

W. S. Luborda
LCDR, USN

In Partial Fulfillment of the Requirements
for the Degree of
Master of Science in Aeronautical Engineering

August 1951

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SUMMARY

To determine the effect of centrifugal force on the forced convection heat transfer coefficient, a rotating test rig was built, housing a 10" length of 1" pipe. The pipe was heated by electrical tubular heating elements. Three thermocouples gave the temperature of entrance and exit air, and the wall. Electrical connections to the heating elements and the thermocouples were made through a slip ring-brush arrangement. Cooling air was drawn in by suction through a hollow shaft in the slip ring attachment and then through the test section.

The results obtained were not satisfactory for any definite conclusions since the test was made under transient thermal conditions and therefore made the heat losses impossible to calculate. The results obtained, however, do tend to suggest that centrifugal force tends to increase the heat transfer coefficient. Modifications to the test rig to establish steady state conditions and means for determining the heat losses must be made prior to reaching any definite conclusions.

INTRODUCTION

The problem of turbine blade cooling is being studied today by many in the aeronautical field, since turbine blade cooling represents the most immediate and positive solution of a high temperature turbine. Reference (a) mentions the fact that analytical methods are too formidable to provide a direct solution, and that the answers to all the factors involved in turbine blade cooling must be derived by experimental research. It was the object of this experiment to determine the effect of centrifugal force on the forced convection heat transfer (surface convection) coefficient, one of the factors involved in the problem.

TEST EQUIPMENT

In order to study the effect of centrifugal force on the surface convection coefficient, it was necessary to design a rotating test rig. The approaches to the design of such an apparatus were many, but it was decided to rotate a length of heated pipe, within and concentric with another larger pipe, which was a primary structural member of the apparatus. The outer pipe provided the strength required to withstand the centrifugal forces due to rotation, while housing the test section. For balance purposes, two such pipes were necessary. In order to rotate the pipes, they were attached to a hub, which in turn was keyed to a shaft. A V-belt drive connected the shaft to an external power source. The addition of heat to the test section and the receiving of temperature measurements from the test section were made electrically by means of slip rings and brushes. The end result of the design is shown in Figure 1. The test rig was controlled from the control room shown in Figure 2. It should be noted that no outside source of air was used, i.e., from compressor or tank. Cooling air was drawn in by suction through the hollow shaft of the slip ring attachment from the atmosphere.

The Test Section

A 1", nominal size, standard steel pipe was used as the test section, having a 1.05" I.D. and a 1.315" O.D. To heat the pipe, five "Chromalox" Inconel sheath, 0.344" diameter, tubular heating elements were used. Each element was rated 280 watts at 115 volts, with a maximum safe sheath temperature of 1400° F. The overall length of an element was 15" with a heated length of 9.5". The latter determined the length of the test section, which was 10". The elements were strapped symmetrically around the test section as shown in Figure 3, in order to provide positive contact between the elements and the test section. The elements were connected in parallel.

Three thermocouples were used to measure the air, at entrance and exit, and the wall. For the air temperature measuring thermocouples, holes were drilled through the housing pipe and the test section for outside insertion. These thermocouples extended into the center of the test section with the entrance air thermocouple $\frac{1}{4}$" in front of the inner end of the test section and the exit air thermocouple $\frac{1}{4}$" inside the outer end. The wall thermocouple was inserted through a hole in the outer pipe, midway between the other two, and

rested against the wall of the test section. The thermocouple wires fitted into a ceramic sleeve which extended into the test section. Figure 4 is a detailed drawing of the test section.

The Test Rig

The hub was made from a solid piece of SAE 4640 steel bar stock, 10" diameter, 8" long. The bar was hollowed out to a depth of 4.75" and turned down to 9.4", leaving a 2" thick rim, 5.5" wide. At the shaft end, the hub was turned down to 3.5" with a 2" fillet. A 1.5" hole was drilled out for the drill rod shaft. The hub was connected to the shaft by two ½" Allen head set screws fitting into holes drilled 120 degrees apart through the hub and into the shaft.

To accomodate the housing pipes, two 3.5" holes were drilled diametrically opposite each other through the rim of the hub. On the inner portion of the rim, 4.5" cut outs were made, concentric with the 3.5" holes, in order to present a flat surface for the holding screw caps. The housing pipes were 3" nominal size, extra strong, steel pipes, having a 2.9" I.D. and a 3.5" O.D. They were 27" long. The inside ends of the pipes, those fitting into the hub, were threaded on the

outside with 2" of 10/inch Acme threads; the outside ends had 2" of 10/inch V threads.

Four special holding screw caps were made. Two had Acme threads, and two had V threads. They were made from 4.25" round steel bar stock. Each cap was 4" indiameter, 2" long, with 1" of inside threads. The other 1" of length provided a ⅛" undercut for threading purposes and a ½" thick wall at the end of the cap. Through this wall, a 1.05" hole was drilled to conform to the test section diameter. The caps were used as centering devices by turning down the wall from the inside to 1.25" diameter for a ¼" depth, concentric with the 1.05" hole.

The Test Arm

The test arm is the pipe housing the test section. It is shown in Figure 5. Since the test section was only 10" long, it was necessary to provide some means of centering it and holding it in place. This was done by using short lengths of 1" pipe and ceramic tubing. The ceramic tubing was a machineable ceramic, called Alsimag 222, having a compressive strength of 25,000 psi and a thermal conductivity of .005 g-cal x cm thick / cm^2 x sec x $^{\circ}$C. The ceramic insulation

at the ends of the test section was built up from 3 tube sizes available. They were cemented together with porcelain cement, so that a tube with a 1.05" I.D., 2.9" O.D., and 1.5" long insulated the test section at each end. Five holes of 0.344" diameter were drilled out to allow the cold ends of the heating elements to pass through, as shown in Figure 3. The photograph also shows how the ends of the test section were turned down to 1.25" O.D. for a ¼" length. These ends fitted into ¼" deep shoulders in the ceramic tubing, similar to the shoulders in each cap. Completing the cover between the two end ceramic tubes was one tube 8" long with a 2.5" I.D. and a 2.9" O.D.

In order to place the test section as far out as possible from the center of rotation, but still contain it within the housing, a short 4.25" section of 1" pipe was used, turned down at the ends to fit into the shoulders of the cap and the ceramic tube. On this short section of pipe, a bumper was built. This consisted of a 2.5" diameter circular steel ring, 3/16" wide, welded to the pipe. On the side of the terminal connections, a ¼" wide ring of fiber board insulated the terminal ends from the steel bumper. The purpose of the bumper was to prevent the outward movement of the heating elements during rotation.

When the caps were screwed on the ends of the housing pipes, the overall length was 28", with 27.5" between the cap shoulders. With a 10" test section, 2" of ceramic tubing, and a 4.25" length of pipe, another 11.25" length of pipe was used to cover the distance between the cap shoulders. Thus, a continuous channel of 1.05" diameter was formed from the end of the inner screw cap to the end of the outer screw cap. When mounted in the hub, the entrance to the test section was located 13.2" from the center of rotation.

Assembly of the Test Arm

Starting with the test section as shown in Figure 3, the test section was covered with the ceramic tubing. The terminal ends of the heating elements were then connected in parallel by ring connectors which were connected together by short strips of No. 12 wire. The terminal ends were bent outward from the 1" pipe toward the outer pipe, so that the terminal ends and their connections formed a ring midway between the two pipes. Two No. 12 insulated wires were connected to a terminal at each end. The two short sections of pipe were then inserted into the ceramic shoulders. A hole to pass the No. 12 wire was drilled in each end of the test arm. The wires were led through these holes to the outside as the test

section was inserted into the test arm.

The Balance Arm

The balance arm is the pipe housing the balance rod. These are shown in Figure 6. The balance rod consisted of a standard 1" pipe, 27½" long, threaded on the surface. Two nuts locked the balancing weight, a brass tube, between them. Threading on the pipe allowed fine adjustment of the location of the balancing weight. The balance rod was turned down on both ends to fit into the cap shoulders.

The Slip Ring Attachment

Figure 7 shows the slip rings and brushes in position. The brass slip rings were mounted on a flanged, hollow shaft, extending outward from the hub. The flange fitted into a recess in the outer face of the hub and was attached to the hub by four ¼" bolts. To insulate the rings from the shaft, a flanged fiber sleeve was made, with the flange end of the sleeve resting against the flange of the shaft. Then, alternately, brass rings and fiber rings were slipped over the fiber sleeve. To keep the rings from turning, the end of the shaft was threaded and a brass nut made, which forced the rings together

when tightened with a spanner wrench.

The dimensions of the slip ring attachment are shown in Figure 8. The electrical connections were made through the hollow shaft. For the two power leads, No. 12 insulated wire was used. On the first ring from the flange, for example, a hole large enough for the bare wire to pass was drilled through the brass ring, fiber sleeve, and the hollow shaft. The brass ring was then removed, and a larger hole was drilled through the fiber sleeve and the shaft to pass the insulated wire. The insulation was removed from the tip of the wire. The brass ring was then slipped back into position, the holes matched, and the bare wire inserted into the hole of the brass ring and soldered to it. The wire was inserted from inside the shaft. The thermocouple wires were fiber glass insulated and the same procedure used in connecting them to the rings. When drilling the holes, the shaft was rotated for each connection, so that when rotating at high speed, the wires would lie against the shaft and not upon one another.

The brushes used were Kimble electro-graphite brushes, No. 512. The brush holder consisted of a piece of fiber board with holes drilled out to fit the brushes loosely, and spaced to contact the brass rings. Brush tension was achieved in two

ways. The brushes were separately spring loaded, with 1.25" long No. 2 compression springs. These were inserted into holes drilled into a plexiglass back plate, which was bolted to the fiber board. The brush holder was mounted on a channel section, which was bolted to the movable portion of a test stand placed in front of the test rig. Thus the brushes could be raised and lowered at will.

Mounting the Test Rig

Four stands were used in mounting the test rig. The shaft was mounted in four Fafnir, LAK type, pillow blocks, containing self-aligning bearings. Two pillow blocks were bolted to the front plate and two on the back plate. The plates were 2" plates, 12" x 18". Each plate was bolted to two stands, each stand bolted to the floor. Between the plates, the shaft was keyed for a quadruple Browning V belt sheave with a 9.4" pitch diameter. On the back plate, a Westinghouse tachometer generator was mounted and connected to the shaft by a belt drive.

Assembly of the Test Rig

After the shaft was mounted in the pillow blocks, the hub was mounted on the end of the shaft. The test arm was passed through the hole in the hub rim and the inner holding cap

screwed on tightly. The test arm was then pulled out so that the cap rested on the flat surface of the inner rim. A 1/8" set screw on the outer face of the hub rim held the pipe in place. Care was taken to see that the 1" pipe ends fitted into the cap shoulders. The balance arm was mounted in the same way. Next, the wires from the slip ring attachment were threaded through a $\frac{1}{2}$" hole in the hub rim, and the flange of the slip ring attachment was bolted to the rim. The wiring and insertion of thermocouples was then completed. The brush holder was mounted on its stand, and raised to make contact with the slip rings.

Motive Power

The engine providing the power was an Onan, CK/SA type, rated 10 hp at 3000 rpm. It was mounted on two 2" x 5" channel irons bolted to the floor. Slots were cut in the top flanges of the channel iron, so that the proper tension on the belts could be obtained by moving the engine and then bolting it down. A quadruple sheave with a 3" pitch diameter was mounted on the engine shaft. The engine was started by using an aircraft inertia starter.

The Control Room

In the control room were located the following: an ignition switch and throttle for the engine, shaft RPM indicator, Brown Recorder for temperature readings, a thermocouple selector switch, a powerstat, voltmeter, ammeter, and a stop watch.

The power for heating the test section was taken from a wall socket, passed through the powerstat, voltmeter and ammeter to the brushes. The powerstat was rated at 230 volts, 9 amperes.

The leads from the thermocouple brushes were passed through the selector switch and on to the Brown Recorder.

The throttle control was a Sperry hydraulic control. The engine had to be started in the test cell, using an aircraft inertia starter, but could be controlled from the control room.

PROCEDURE

In checking out the equipment, it was found that the maximum pipe temperature that could be maintained at high RPM was approximately 400° F., with full power from the powerstat. Therefore, the following procedure was used. The engine was run at low speeds while the test section heated up. When the test section temperature became 700° F., the stop watch was started and the temperatures recorded as of zero time. The throttle was then advanced to the next RPM and allowed to run for three minutes, at which time the temperatures were again recorded, and the throttle advanced to the next RPM. The electrical power input to the test section was left constant. This procedure was followed until the maximum RPM was reached.

RESULTS AND DISCUSSION

Three runs were made. The first two runs, shown in Tables I and II, were trial runs to observe the operation of the equipment, with and without heat added, respectively. Therefore, no times were recorded. Readings were taken when conditions seemed steady. These runs are recorded merely to indicate the trend of the readings. The low temperatures recorded in the "no heat" run are of interest.

The data for the test run is given in Table III, and the reduced data in Table IV. The test run was a double run, i.e., data was taken at increasing RPM's until the maximum of 800 RPM was reached and then data for decreasing RPM's was taken.

The reduced data in Table IV was computed assuming no heat losses from the test section. With power maintained constant at 810 watts, the wall temperature decreased as the RPM increased. Thus, the aspirated air flowing through the test section not only absorbed the power input but also cooled the pipe. The timing between RPM settings allowed calculation of this added heat absorption by the air stream. Thus, knowing the rate of heat flow and the difference in temperature

between the wall and main body of the air, the heat transfer (surface convection) coefficient was computed from the formula

$$q = h A_s (T_p - T_a) \tag{1}$$

where

q = rate of heat flow, Btu/hr

A_s = area of the surface, ft^2

h = heat transfer coefficient, Btu/hr ft^2 °F

T_p = temperature of the wall, °F

T_a = average temperature of the main body of air, °F

The centrifugal force acting on the air particles flowing through the test section could not be computed since only one property of the cooling air was known, i.e., the temperature. However, centrifugal force is a function of the rotational speed. Thus, the values of the surface convection coefficient, as determined by equation (1), were plotted in Figure 9 against rotational speed.

Now, in order to make a comparison between the heat transfer coefficient obtained with rotation (due to centrifugal force) and the heat transfer coefficient due to forced convection only, the following was done.

From the data taken, the weight flow was computed from the formula:

$$q = w\ c_p\ (\Delta T) \qquad (2)$$

where

q = rate of heat flow, Btu/hr

w = weight flow, lb/hr

c_p = specific heat of air, Btu/lb °F

ΔT = temperature difference between leaving and entering air, °F.

The computed values of w for each RPM are given in Table IV. From the continuity equation, the mass velocity of the air was computed,

$$w = \rho uA = GA \qquad (3)$$

where

G = (ρu) mass velocity, lb/hr ft^2

A = area of test section, ft^2

Using the values of G as computed for each weight flow at each RPM, the surface convection coefficient due to forced convection was computed from the simplified formula

$$h = C\ \frac{G^{0.8}}{D^{0.2}} \qquad (4)$$

where

C = constant

D = diameter of test section, ft

G = mass velocity, lb/hr ft^2

Equation (4) is given as equation (6-8) of reference (b), in which values for the constant C for air at various temperatures are given in Table 6-3.

Since the heat transfer coefficients computed by equation (4) were based on the weight flow from the test data at each RPM, they were also plotted in Figure 9. Thus, Figure 9 should be read as follows: Each RPM indicates a certain weight flow. This weight flow through a static rig would have a heat transfer coefficient indicated by the forced convection curve. In a rotating rig, this same weight flow would have a heat transfer coefficient indicated by the centrifugal force curve. The weight flows for each RPM are given in Table IV. Thus, for a given weight flow of cooling air, centrifugal force tends to increase the heat transfer coefficient.

Cognizance must be taken of the fact that only a short range of rotational speeds were covered in the test runs. In addition, the computations were based on the assumption that no losses existed. These factors and the inaccuracies in reading the Brown Recorder, with marked intervals of 20°, cannot allow any quantitative results. Thus, the straight lines drawn in Figure 9 are not necessarily so.

The test run was made under transient thermal condi-

tions, as indicated by the test data. This was done purposely since the time required to reach stable conditions at each RPM setting was too great, e.g., the wall temperature of the pipe was increasing at the rate of 10° F. per minute after 18 minutes of warm-up time when zero time was recorded.

The transient thermal conditions thus made the calculation of heat losses impossible. In the Sample Calculations section, heat losses were calculated from the test section to the outside air through the air cylinder surrounding the test section. The heat losses calculated are very low, because they were based on a stagnant air cylinder. Circulation of the air within this cylinder would greatly increase the heat loss rate.

An indication of the magnitude of the heat losses are attested to by the high inlet air temperatures. These air temperatures show that the air was being heated prior to entering the test section. At the slow RPM and the consequent low weight flow, the air inlet temperature reached 340° F. in 18 minutes from 80° F. atmospheric air. However, at 800 RPM the inlet air temperature was recorded as 60° F. When the rotational speed was decreased from 800 RPM to 450 RPM, an elapsed time of 21 minutes, the inlet air temperature was only 95° F., or a 15° ~~95°~~ F. increase above atmospheric air when flowing through

the first 11½" of pipe ahead of the test section. At the same time, the difference in the temperature between entrance and exit air was 175° F. Thus the heat lost to pipe ahead of the test section is roughly 10% of the heat added to the test section. In addition to this loss, a further loss to the pipe aft of the test section exists.

All these losses decreased the power input to the test section. Consequently, the heat transfer coefficient due to centrifugal force as computed by equation (1) are too high.

From the test data taken, the results tend to suggest that centrifugal increases the heat transfer coefficient. Modifications to the test section will be required in order to calculate the heat losses. These modifications should be such as to enable the operator to establish steady conditions.

As a matter of interest, the Reynold's Number for both heat transfer coefficients was computed from the forced convection formula

$$h = .0202 \frac{k}{D} (N_R)^{0.8} \qquad (5)$$

where k for air was computed from Eucken's formula

$$k = k_{32} \frac{492 + C}{T + C} \left(\frac{T}{492}\right)^{3/2} \qquad (6)$$

Results are tabulated in Table V. For a given weight flow, the Reynolds number as computed for the centrifugal force heat transfer coefficient is a fictitious value and is designated as N_R^1.

CONCLUSIONS AND RECOMMENDATIONS

The test run made and the results obtained tend to suggest that centrifugal force will increase the heat transfer coefficient. Verification of this and the amount of increase must be established by further testing.

In making future tests, the following changes to the test rig are recommended:

1. Replace engine used for motive power with a more powerful engine to attain higher and lower rotational speeds.

2. Use a more accurate RPM indicator.

3. Use a higher rated powerstat in order to be able to stabilize conditions and determine accurately the power absorbed by the cooling air.

4. Devise means for accurately determining heat losses under steady state conditions.

5. Use separate potentiometers for each thermocouple for more accurate temperature readings.

6. With the test arm horizontal, supply cooling air from an outside source for forced convection. Duplicate these

results with centrifugal force.

7. Use a test section with a smooth inner surface and compare results of tests on it and the test section used in this experiment.

8. Devise a method to install an orifice meter at the hollow shaft of the slip ring attachment to accurately determine the weight flow passing through the test section and thus check data results.

FORMULAS AND SAMPLE COMPUTATIONS

From line 5, Table IV:

Eq. (1) $q = h_{cc} A_s (T_p - T_a)$

$$q = P_{tot} \text{ watts} \times \frac{3.412 \text{ Btu/hr}}{\text{watt}}$$

$$P_{tot} = P_p + \Delta P_p$$

where

P_p = power input to heat pipe, watts

ΔP_p = power loss in pipe due to temperature decrease, watts

$$P_p = 90\text{v} \times 9\text{a} = 810 \text{ watts}$$

$$\Delta P_p = \frac{\text{wt of pipe (lbs)} \times \text{sp ht} \times {}^{\circ}\text{F drop}}{3.412 \times \text{time, hrs.}}$$

$$= \frac{1.689 \text{ lbs/ft} \times \frac{10}{12} \text{ ft} \times .12 \text{ Btu/lb } {}^{\circ}\text{F} \times 45}{3.412 \times .05 \text{ hrs}}$$

$$\Delta P_p = 44.6 \text{ watts}$$

$$\therefore P_{tot} = 810 + 44.6 = 854.6 \text{ watts}$$

$$q = 854.6 \text{ watts} \times \frac{3.412 \text{ Btu/hr}}{\text{watt}}$$

$$q = 2915 \text{ Btu/hr}$$

$$T_a = \frac{T_1 + T_5}{2}$$

$$T_a = \frac{210 + 85}{2}$$

$$T_a = 147.5^{\circ} \text{ F}$$

$$h_{cc} = A_s \left(\frac{q}{T_p - T_a}\right)$$

$$h_{cc} = \frac{2915 \text{ Btu/hr}}{\pi\left(\frac{1.05}{12}\right)\left(\frac{10}{12}\right) \text{FT}^2 \; (485 - 147.5) \; {}^{\circ}\text{F}}$$

$$h_{cc} = \frac{2915}{.229 \times 337.5} = 37.8 \text{ Btu/hr ft}^2 \; {}^{\circ}\text{F}$$

Equation (2)

$$w = \frac{q}{C_p \, \Delta T}$$

$$\Delta T = T_1 - T_3$$

$$\Delta T = 210 - 85 = 125^{\circ} \text{ F}$$

$$w = \frac{2915 \text{ Btu/hr}}{.24 \text{ Btu/lb } {}^{\circ}\text{F} \times 125^{\circ} \text{ F}}$$

$$w = 97.1 \text{ lb/hr}$$

Equation (4)

$$h_{cf} = C \, \frac{G^{0.8}}{D^{0.2}}$$

$$G = \rho V = \frac{w}{A} = \frac{97.1 \text{ lb/hr}}{\frac{\pi}{4} \; \frac{1.05}{12}^{2} \; \text{ft}^2}$$

$$G = \frac{97.1}{.006} = 16{,}200 \text{ lb/hr ft}^2$$

C = .00346 for air at 140° F, Table 6-3, ref. (a)

$$D^{0.2} = 0.614$$

$$h_{cf} = .00346 \; \frac{(16{,}200)^{0.8}}{.614} = \frac{.00346 \; (2320)}{.614}$$

$$h_{cf} = 13.1 \text{ Btu/hr ft}^2 \; {}^{\circ}\text{F}$$

Equation (6)

$$k = k_{32} \, \frac{492 + C}{T + C} \left(\frac{T}{492}\right) 3/2 \text{ Btu/hr ft}^2 \; {}^{\circ}\text{F/ft}$$

For air: $k_{32} = 0.0125$

$C = 225$

$T = (147.5 + 460)\ ^{\circ}\text{F abs}$

$$k = (.0125)\left(\frac{492 + 225}{607.5 + 225}\right)\left(\frac{607.5}{492}\right)^{3/2}$$

$$k = (.0125)\left(\frac{717}{832.5}\right)\ (1.37)$$

$$k = .0147$$

Equation (5)

$$h_{cf} = .0202\ \frac{k}{D}\ (N_R)^{0.8}$$

$$(N_R)^{0.8} = \frac{h_{cf} \times D}{.0202 \times k}$$

$$N_R = \left[\frac{(13.1) \times \left(\frac{1.05}{12}\right)}{.0202 \times .0147}\right]^{1.25}$$

$$N_R = \left[3860\right]^{1.25}$$

$$N_R = 30{,}100$$

Heat loss:

Compute radial heat loss from test section to outside air, assuming the stagnant air between the test section and test arm as the only barrier.

$$q_1 = \frac{2\pi k_m\ (t_1 - t_2)}{\log_e\ (r_2/r_1)}$$

t_1 = temperature of inside surface (temp. of wall)

t_2 = temperature of outside surface (atmospheric air temp)

r_1 and r_2 = inner and outer radii of air

The air cylinder was assumed as 2", inner diameter and 2.9" outer diameter. Thus, the losses are exaggerated.

For 800 RPM:

$t_1 = 410^\circ$ F

$t_2 = 80^\circ$ F

k_m = .0186 Btu/°F hr ft^2/ft (for air)

$$q_1 = \frac{2\pi(.0186)(410-80)}{\log_e \frac{1.45}{1.00}} = \frac{2\pi(.0186)(310)}{.372} \text{ Btu/hr ft}$$

q_1 = 98 Btu/hr ft

$q_1 = 98 \times \frac{9.5}{12}$ ft = 77.5 Btu/hr

At 450 RPM:

$$q_1 = \frac{2\pi(.0186)(520-80)}{.372}$$

q_1 = 132 Btu/hr ft $\times \frac{9.5}{12}$ ft

q_1 = 104.5 Btu/hr

The log mean temperature difference on two sides of the film:

$$\theta_m = \frac{\theta_a - \theta_b}{\log_e \frac{\theta_a}{\theta_b}}$$

At 300 RPM: $T_2 = 410^\circ$ F $T_1 = 155^\circ$ F $T_3 = 60^\circ$ F

$$\theta_m = \frac{155-60}{\log_e \frac{350}{255}} = \frac{95}{\log_e 1.37}$$

$$\theta_m = \frac{95}{.317} = 300^\circ \text{ F}$$

At 450 RPM: $T_2 = 520^\circ$ F $T_1 = 270^\circ$ F $T_3 = 95^\circ$ F

$$\theta_m = \frac{270-95}{\log_e \frac{425}{250}} = \frac{175}{.53}$$

$$\theta_m = 326$$

Corrected h at 300 RPM:

$$h = \frac{Q-Q_1}{A_s\theta_m} = \frac{2815-77.5}{.229 \times 300}$$

$$h = \frac{2737.5}{.229 \times 300} = 39.8$$

$$\% \text{ error} = \frac{40.5-39.8}{39.8} = \frac{.7}{39.8} = 1.78\% \text{ increasing}$$

Corrected h at 450 RPM

$$h = \frac{Q-Q_1}{A_s\theta_m} = \frac{2700-105}{.229 \times .326}$$

$$h = \frac{2595}{.229 \times 326} = 34.8$$

$$\% \text{ error} = \frac{34.9-34.8}{34.8} = \frac{.1}{34.8} = 0.3\%$$

NOMENCLATURE

A	Area	sq. ft.
A_s	Surface area	sq. ft.
c	Specific heat	Btu/lb ^oF
C	Constant	
D	Diameter	ft
G	Mass velocity	lb/hr ft^2
h_{cc}	Surface coefficient due to centrifugal force	Btu/hr ft^2 ^oF
h_{cf}	Surface coefficient due to forced convection	Btu/hr ft^2 ^oF
k	Thermal conductivity	Btu/hr ft^2 $^oF/ft$
N_R	Reynolds number	
$N_R{}^1$	Pseudo-Reynolds number	
P	Power	watts
q	Rate of heat flow	Btu/hr
T	Temperature	^oF
T_a	Average temperature of air	^oF
T_p	Temperature of test section	^oF
u	Velocity	ft/hr

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APPENDIX

TABLE I

TEST RIG CHECK OUT. HEAT ADDED

RPM	Volts	Amps	T_1	T_2	T_3
300	83.5	8.2	474	700	340
400	89.0	9.0	470	682	225
500	90.0	9.0	310	600	82
600	89.0	9.0	200	550	65
700	89.0	9.0	185	480	60
800	89.0	9.0	145	420	40

TABLE II

TEST RIG CHECK OUT. NO HEAT ADDED

RPM	Volts	Amps	T_1	T_2	T_3
0	0	0	75	75	75
500	0	0	50	45	45
700	0	0	30	25	15
600	0	0	30	20	15
400	0	0	35	20	20
300	0	0	40	25	25
250	0	0	45	38	38

Note: T_1 = Exit Air Temperature, °F

T_2 = Wall Temperature, °F

T_3 = Entrance Air Temperature, °F

TABLE III

TEST RUN

Barometer 29.22" Hg Temperature 80° F

RPM	Time, min.	Volts	Amps	T_1	T_2	T_3
300	0	90	9	560	710	340
400	2	90	9	530	690	300
500	5	90	9	400	645	160
550	8	90	9	310	580	110
600	11	90	9	260	530	100
650	14	90	9	210	485	85
700	17	90	9	200	455	75
750	20	90	9	165	425	60
800	23	90	9	155	410	60
750	26	90	9	160	415	60
700	29	90	9	175	425	62
650	32	90	9	180	435	65
600	35	90	9	200	460	70
550	38	90	9	210	480	78
500	41	90	9	250	500	85
450	44	90	9	270	520	95

400 Arcing at brushes

Note 1. Nomenclature:

T_1 = Exit Air Temperature, °F

T_2 = Wall Temperature, °F

T_3 = Entrance Air Temperature, °F

Note 2. Time required for T_2 to reach 700° F at 300 RPM was 18 minutes.

TABLE IV

REDUCED DATA

N	P	ΔT_p	ΔP_p	P_{tot}	q	T_p-T_a	h_{co}	ΔT	w	C	G	h_{of}
400	810	-33	29.8	839.9	2860	275	45.5	230	51.8	.00362	8,650	8.35
500	810	-45	44.6	854.6	2920	365	34.9	240	50.6	.00349	8.450	7.90
550	810	-65	64.5	874.5	2980	370	35.2	200	62.0	.00343	10,300	9.10
600	810	-50	49.6	859.6	2930	350	36.6	160	76.1	.00342	12,700	10.75
650	810	-45	44.6	854.6	2915	337.5	37.7	125	97.0	.00340	16,200	12.95
700	810	-30	29.8	839.8	2860	317.5	39.4	125	95.5	.00339	15,900	12.70
750	810	-30	29.8	839.8	2860	312.5	40.0	105	113.5	.00337	18,900	14.5
800	810	-15	14.9	824.9	2815	303.5	40.5	95	123.2	.00337	20,600	15.7
750	810	15	-5.0	805.0	2745	305.0	39.4	100	114.5	.00337	19,100	14.68
700	810	10	-9.9	800.1	2730	306.5	38.8	113	99.5	.00337	16,600	13.10
650	810	10	-9.9	800.1	2730	312.5	38.1	115	97.8	.00338	16,300	12.90
600	810	25	-24.8	786.2	2680	325.0	36.0	130	86.5	.00338	14,400	11.75
550	810	20	19.9	790.1	2700	331.0	35.6	132	85.2	.00339	14,200	11.65
500	810	20	19.9	790.1	2700	332.5	35.4	165	68.2	.00340	11,520	9.9
450	810	20	19.9	790.1	2700	337.5	34.9	175	64.4	.00341	10,700	9.34

TABLE V

REYNOLDS NUMBER COMPUTATIONS

N	T	k	h_{cf}	N_R	h_{cc}	N_R^1
800	567	.0139	15.70	40,900	40.5	134,200
750	570	.0141	14.68	36,900	39.4	126,000
700	579	.0142	13.10	31,600	38.8	123,000
650	583	.0143	12.90	30,800	38.1	118,000
600	595	.0145	11.75	27,000	36.0	109,000
550	609	.0148	11.65	26,000	35.6	104,000
500	628	.0152	9.9	20,600	35.4	101,000
450	643	.0153	9.34	18,900	34.9	95,000

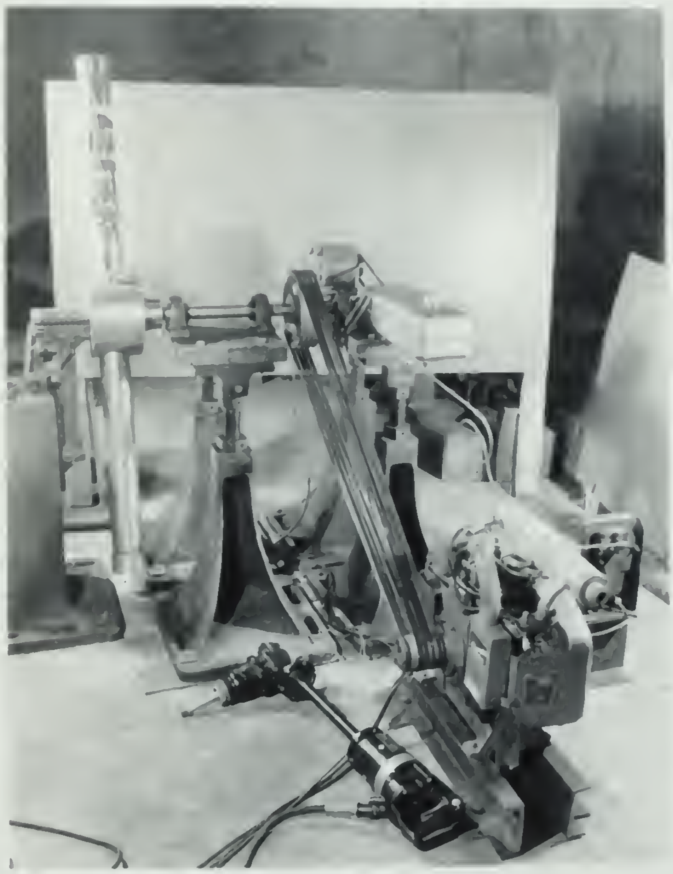

FIG. 1. TEST RIG.

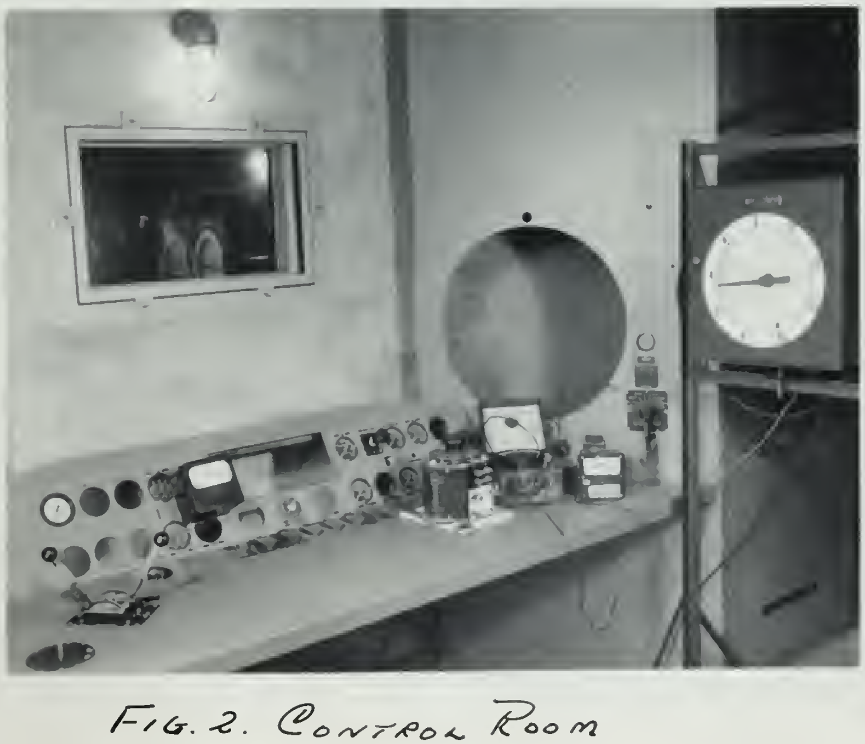

FIG. 2. CONTROL ROOM

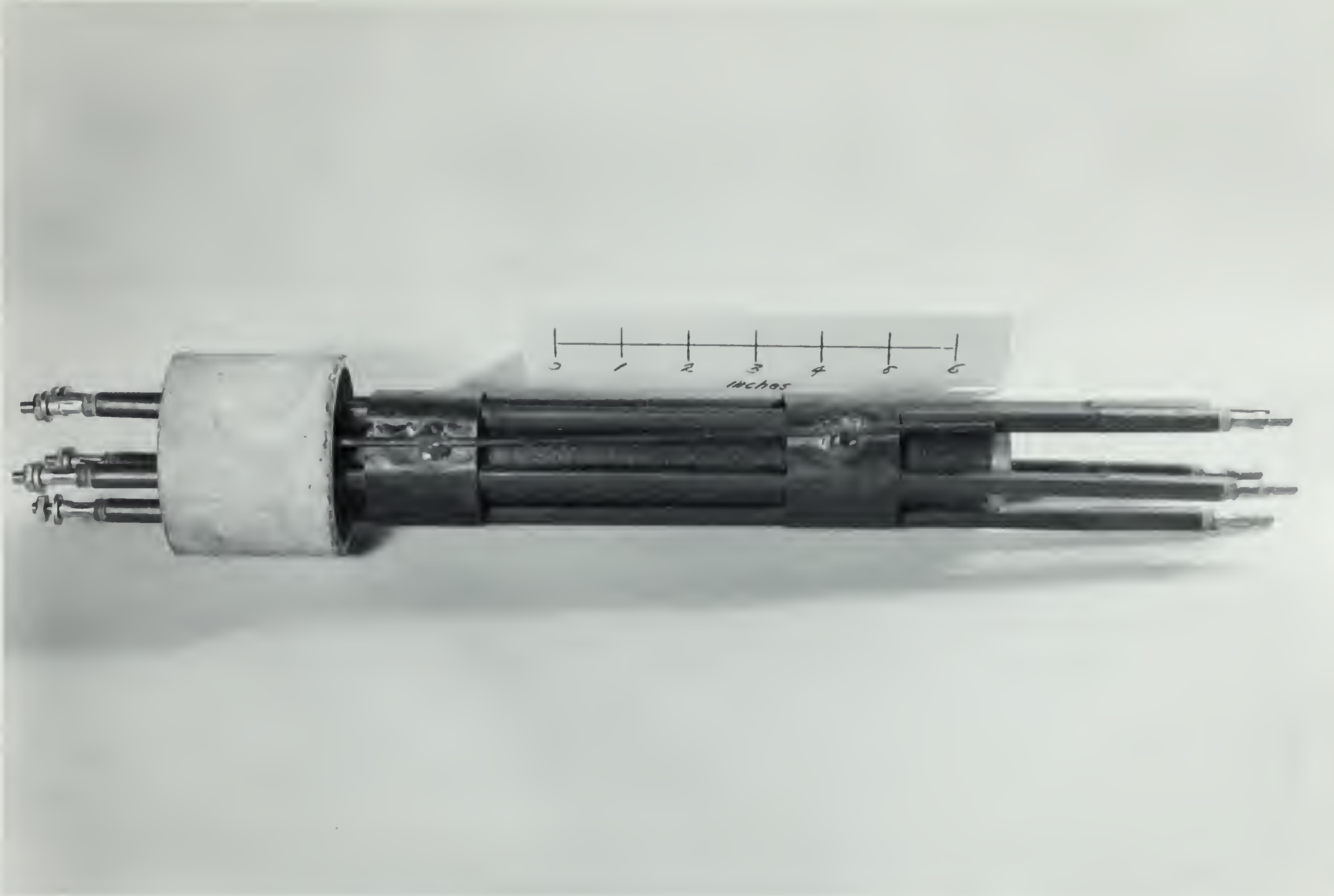
0
1
2
3
4
5
6
inches

FIG. 3. TEST SECTION.

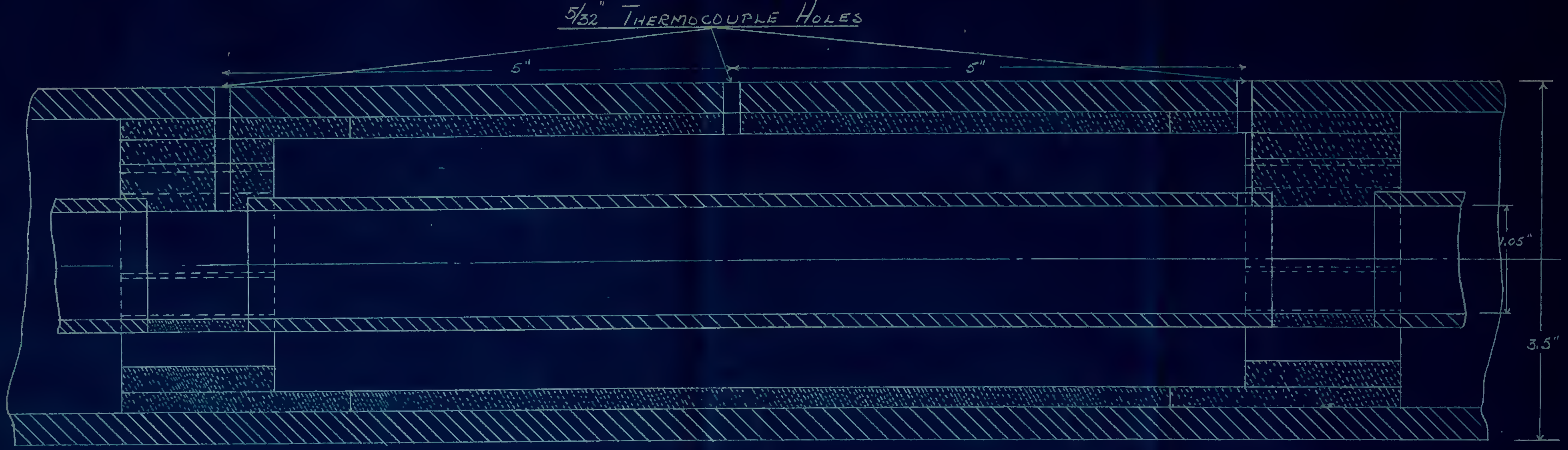
5/32" Thermocouple Holes
5"
5"
1.05"
3.5"

Fig. 4. Cross-Sectional View of Test Section in Test Arm, Showing Ceramic Enclosure.

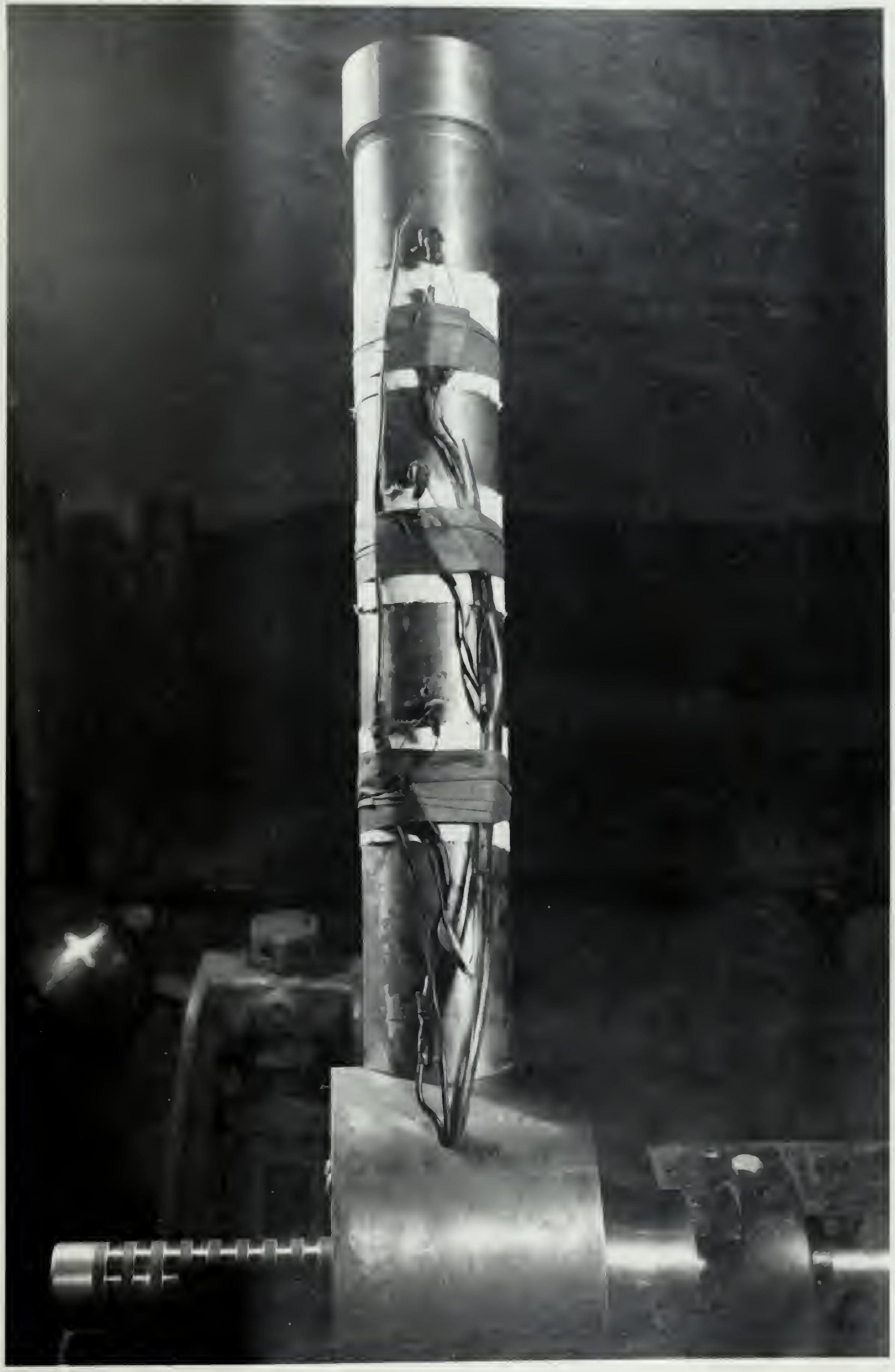

FIG. 5. TEST ARM

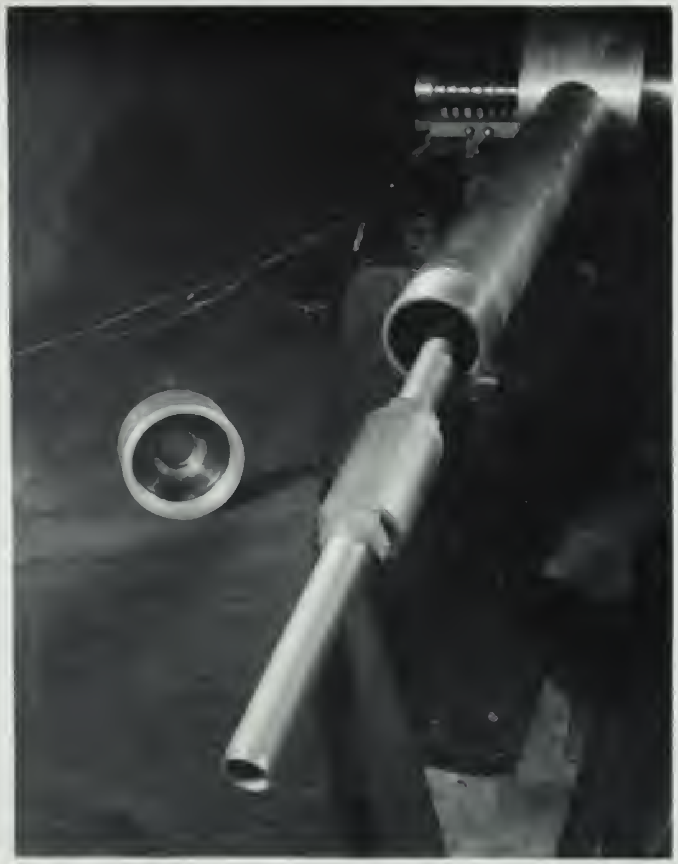

FIG. 6. THE BALANCE ARM.

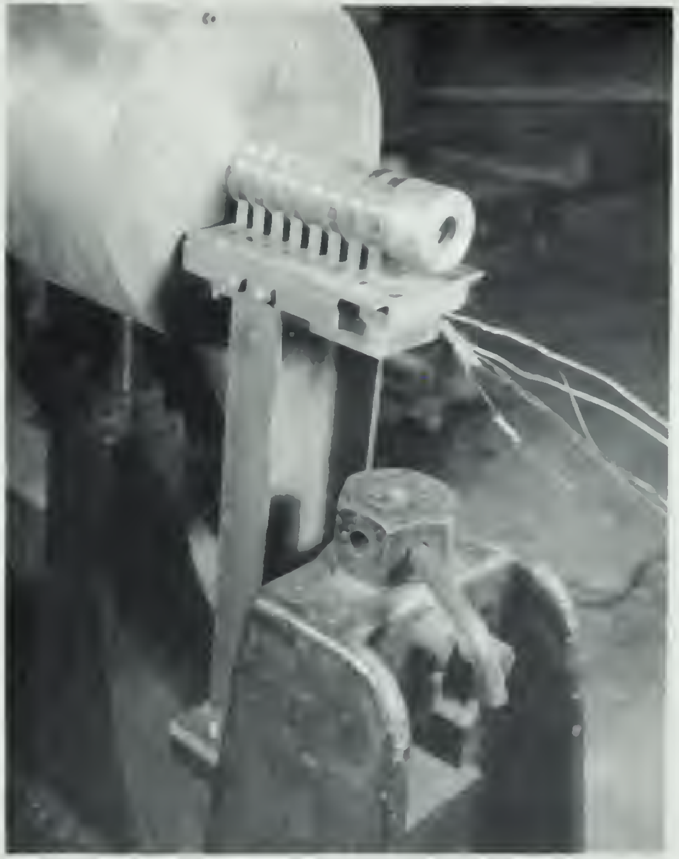

FIG. 7. SLIP RING-BRUSH ARRANGEMENT.

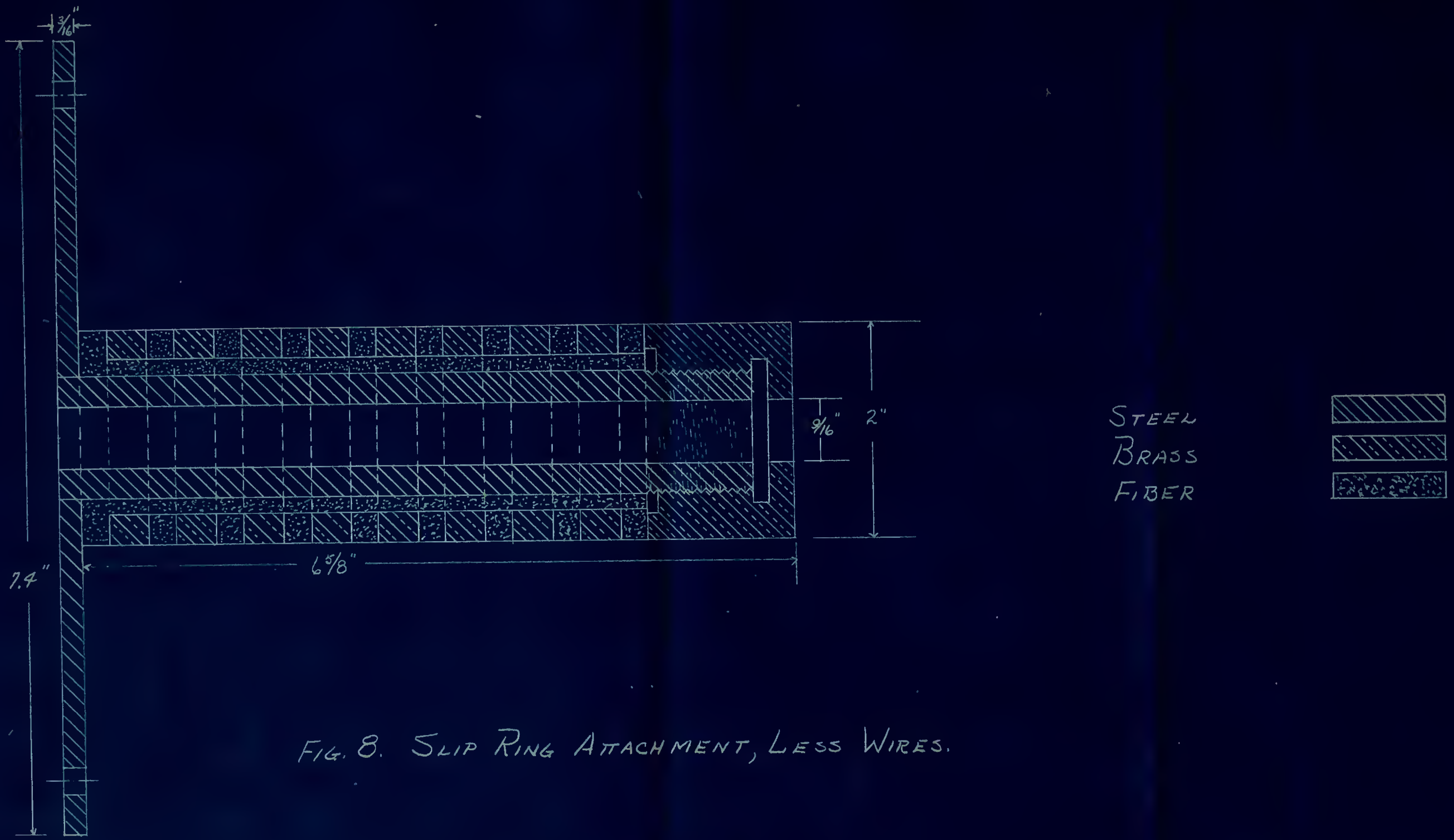
3/16"
7.4"
6 5/8"
9/16"
2"
Steel
Brass
Fiber

Fig. 8. Slip Ring Attachment, Less Wires.

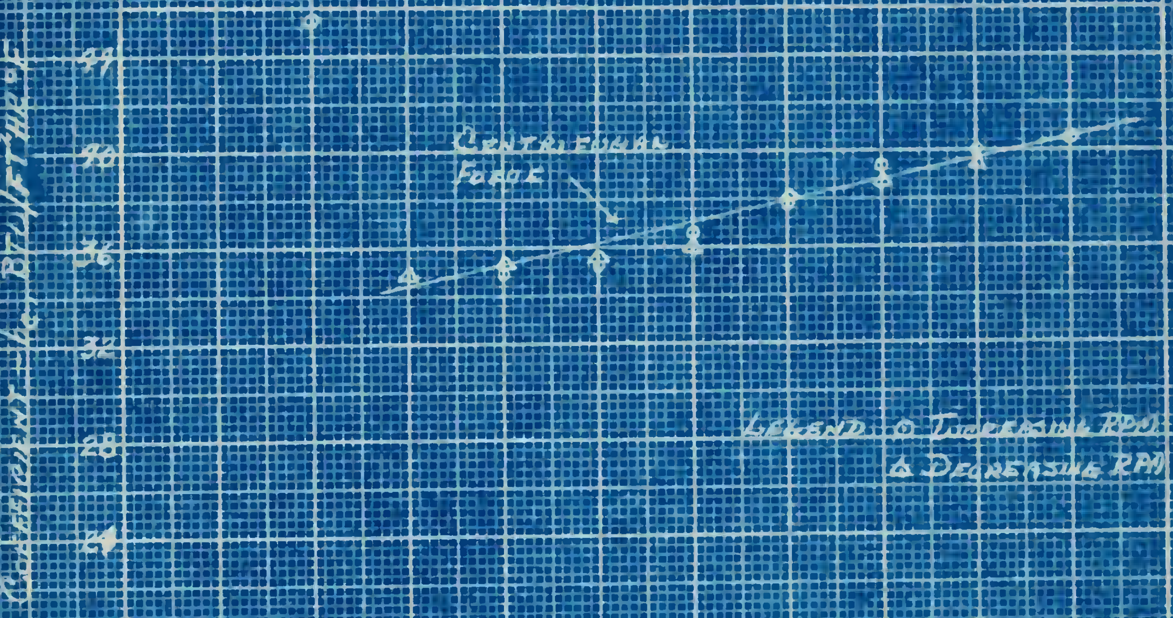

FIG. 9. SURFACE CONVECTION COEFFICIENTS VERSUS ROTATIONAL SPEED

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The effect of centrifugal force on the forced convection heat transfer coefficient.

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